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ABSTRACT

One of the thermodynamic losses in the vapor-compression refrigeration cycle is the throttling process in the expansion valve. If work is extracted from the refrigerant during the expansion process, the efficiency of the cycle is significantly improved. It is proposed that the high-pressure refrigerant be used as the motive fluid of a jet ejector. Instead of extracting mechanical work from the expanding refrigerant, its kinetic energy is used to partially compress the saturated vapor leaving the evaporator, increasing the enthalpy change in the evaporator and reducing the load on the compressor. A first-order analysis of the cycle performance shows significant increase in coefficient of performance and decrease in compressor displacement relative to a standard vapor-compression cycle. The analysis shows much greater performance changes for some refrigerants than for others, indicating a potential impact on the selection of new, non-CFC refrigerants.

NOMENCLATURE

COP	Coefficient of Performance
h	specific enthalpy
P	pressure
r	motive flow/total flow
s	specific entropy
u	velocity
x	quality
η	efficiency

Subscripts:

d	diffuser outlet
fc	condenser saturated liquid
ge	evaporator saturated vapor
m	mixing section outlet
n	motive nozzle outlet
s	suction nozzle outlet

INTRODUCTION

Thermodynamic Losses of the Vapor-Compression Cycle

The basic vapor-compression refrigeration cycle has five major thermodynamic losses which reduce its COP below that of a Carnot cycle:

1. Heat exchange across a temperature difference in the evaporator.
2. Heat exchange across a temperature difference in the condenser.
3. Compressor inefficiency.
4. Heat exchange from superheated vapor at the compressor discharge.
5. Throttling process in the expansion valve.

The first three of these losses are functions of the equipment used to implement the cycle. The last two, however, are intrinsic losses of the cycle. An idealized vapor-compression cycle, in which the first three losses are eliminated, still has the last two losses reducing its COP. The size of these two losses varies from refrigerant to refrigerant.

This paper addresses a means for reducing the loss due to the throttling process in the expansion valve.

Use of an Ejector as a Refrigerant Expander

It has long been recognized that the COP of the vapor-compression cycle would be improved by replacing the expansion valve with some sort of work-producing device, changing the isenthalpic process to an essentially isentropic one. This change would give two benefits: it would reduce the enthalpy of the refrigerant entering the evaporator, and it would provide work to help power the compressor. The work-producing device could be a reciprocating, rotary, or turbine expander, but such a device would be expensive and prone to damage by low quality two-phase flow.

The jet ejector is low in cost and able to handle a wide range of multi-phase flows without damage. It is proposed that an ejector be used as a refrigerant expander. The "ejector expansion" refrigeration cycle is shown in Figure 1. The high-pressure liquid leaving the condenser is used as the ejector motive fluid, partially compressing the saturated vapor leaving the evaporator. A two-phase flow leaves the ejector at a pressure between the evaporator pressure and the compressor discharge pressure. The liquid portion of this flow is returned to the evaporator, while the vapor portion enters the compressor suction. In essence, the result is a two stage refrigeration system, with the work otherwise lost in the high stage expansion process providing the work input for the low stage. The low stage throttling process is across a small pressure difference and thus causes little loss.

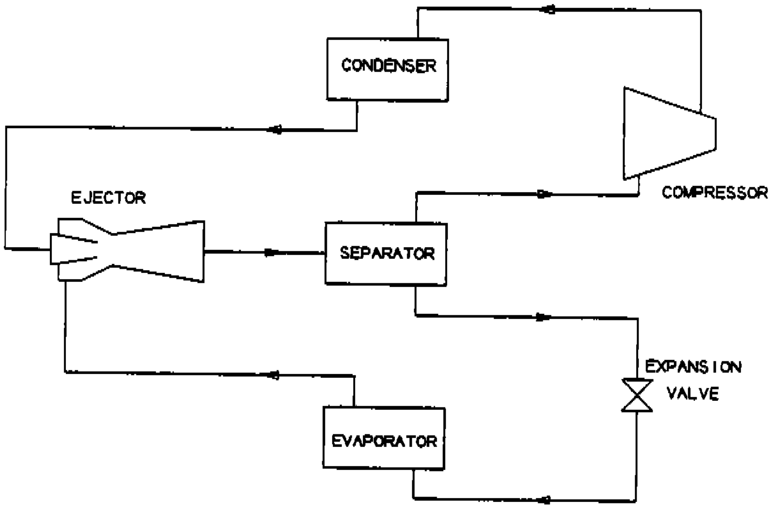

Figure 1 - Schematic of Ejector Expansion Refrigeration System.

The concept was patented by Kemper et al in 1966. Additional development was done at the York Division of Borg-Warner Corporation, resulting in two patents relating to controls for the cycle (Newton, 1972a, 1972b). As far as the author knows, this paper reports the first work on the subject since that time.

ANALYSIS

In order to compare the performance of the ejector expansion refrigeration cycle with the standard vapor-compression cycle, simulations of the two cycles were carried out for the same evaporator temperatures, condenser temperatures, compressor efficiencies, and heat loads. Those components which were common to the standard and ejector expansion cycles were modeled as ideal elements. More attention was given to the modeling of the ejector.

The operation of ejectors has been extensively studied but is not fully understood. Keenan et al (1950) presented the definitive analysis of single-phase ejectors using ideal gases. Stoeker (1958) gave analyses of ejectors using superheated or high quality steam. Flügel (1941) presented analyses of ejectors with various combination of gas, liquid, and two-phase motive and suction fluids. Bonnington and King (1972) presented an extensive bibliography of ejector research.

Most of the studies described above modeled ejectors in an essentially one-dimensional fashion. They used three ways of modeling the ejector: with mixing at constant pressure, with mixing at constant area, and with a combination of constant pressure and constant area mixing. In this analysis constant pressure mixing was assumed, along with other ejector and system assumptions listed below.

Assumptions

The analysis assumed that:

1. Ejector mixing section was shaped so that mixing took place at constant pressure. This pressure could be below evaporator pressure.
2. Except for during the ejector mixing process, properties and velocities were constant over cross sections. The analysis was thus one-dimensional.
3. The refrigerant was at all times in thermodynamic quasi-equilibrium. Together with assumption 2, this corresponded to what is known in two-phase flow as a "homogenous equilibrium model."
4. Processes in ejector nozzles and diffuser and in the compressor were such that deviations from adiabatic reversible processes could be expressed in terms of efficiencies. Any shock effects were included in these efficiencies.
5. Pressure drop in piping, evaporator, and condenser was negligible. Kinetic energy was negligible outside the ejector.
6. No heat transfer to the environment took place except in the evaporator and condenser.
7. Refrigerant leaving the evaporator or condenser was saturated vapor or liquid respectively, and the liquid-vapor separator was 100% efficient.

The limitations imposed by these assumptions will be discussed later.

Calculation Procedure

Mixing pressure, at some value below evaporator pressure, was selected. An initial ratio of motive flow to total ejector flow, r, was assumed. At the outlet of the motive nozzle, by conservation of energy:

$$h_n = (1 - \eta_n) h_{fc} + \eta_n h(s_{fc}, P_m), \text{ and} \tag{1}$$

$$u_n = \sqrt{2(h_{fc} - h_n)}. \tag{2}$$

At the outlet of the suction nozzle, by conservation of energy:

$$h_s = (1 - \eta_s) h_{ge} + \eta_s h(s_{ge}, P_m), \text{ and} \tag{3}$$

$$u_s = \sqrt{2(h_{ge} - h_s)}. \tag{4}$$

At the outlet of the mixing section, by conservation of energy and momentum:

$$u_m = (1 - r) u_s + r u_n, \tag{5}$$

$$h_m = (1 - r) h_{ge} + r h_{fc} - \frac{u_m^2}{2}, \text{ and} \tag{6}$$

$$s_m = s(h_m, P_m). \tag{7}$$

At the outlet of the diffuser, by conservation of energy:

$$h_d = h_m + \frac{u_m^2}{2}, \tag{8}$$

$$h'_d = h_m + \eta_d \frac{u_m^2}{2}, \tag{9}$$

$$P_d = P(s_m, h'_d), \text{ and} \tag{10}$$

$$r = x_d = x(P_d, h_d). \tag{11}$$

The calculated value of r was compared with the value assumed at the beginning of the analysis and a new value was chosen. It was found that if the arithmetic mean of the two values was used, the solution converged rapidly.

Once the ejector performance had been calculated, the performance of other cycle components was calculated in ordinary fashion. Performance was also calculated for the equivalent standard cycle. COP and compressor displacement for the ejector expansion and standard cycles could then be compared.

A Fortran computer program was used to perform the calculations. In order to automate the calculation process, refrigerant properties were calculated from equations of state. Properties for halocarbon refrigerants were calculated using routines based on Downing (1974). Properties for ammonia were calculated using routines based on Haar and Gallagher (1978).

Selection of Mixing Pressure

Calculations were initially performed with mixing taking place at evaporator pressure, then lower pressures were tried. It was found that, for given operating conditions and nozzle and diffuser efficiencies, there was a mixing pressure below suction pressure that gave optimum ejector performance and cycle COP.

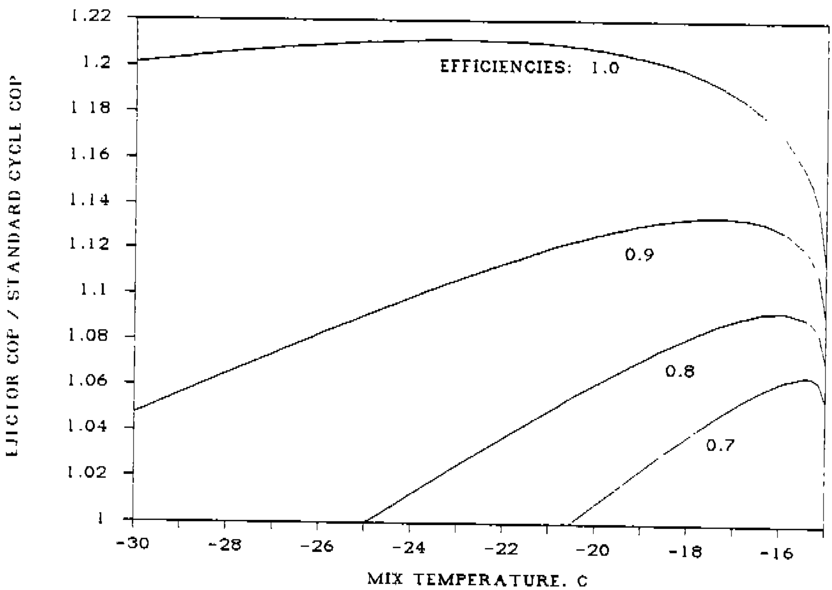

Figure 2. COP Ejector Expansion / COP Standard Cycle vs Mixing Temperature. R-12; -15 C (5 F) Evaporator Temperature; 30 C (86 F) Condenser Temperature; Compressor Efficiency 1, other Efficiencies equal as given.

An explanation of this is found by examining a temperature-entropy or pressure-enthalpy diagram for a typical refrigerant. In the low quality two-phase region lines of constant enthalpy are nearly parallel to lines of constant entropy. In the high quality two-phase region, however, the lines have considerably different slope. Expansion through an isentropic nozzle across the same pressure difference results in greater enthalpy change, and thus in greater velocity, for a high quality fluid than for a low quality fluid. In the two-phase ejector the low quality motive fluid expands across a large pressure difference, while the high quality suction fluid expands across a small pressure difference. The mixing pressure can, in fact, be selected so that the two fluids have the same velocity when entering the mixing section. The loss due to mixing streams of different velocities is eliminated. In cases where both fluids are two-phase at the nozzle outlets they are also at the same temperature, and mixing becomes a reversible process.

Figure 2 shows the COP improvement vs mixing temperature (saturation temperature corresponding to mixing pressure) for a single operating condition and various efficiencies. For efficiencies of 1.0, the maximum of the curve represents the mixing temperature at which suction and motive fluids attain equal velocities. At higher mixing temperatures the motive fluid is moving faster, while at lower temperatures the suction fluid is moving faster.

For an ejector with nozzle and diffuser inefficiencies, use of the optimum mixing pressure based on efficiencies of 1.0 results in large losses in the expansion and recompression of the fluid streams. The selection of optimum mixing pressure becomes a tradeoff between mixing loss and nozzle and diffuser losses. The optimum mixing temperature moves closer to evaporator temperature as efficiencies decrease.

The program written to calculate cycle performance for a given mixing pressure was modified to optimize mixing pressure for each set of conditions. All performance data presented in the Results section is at optimum mixing pressure for the stated conditions.

Limitations of Analysis

The one-dimensionality of this analysis is intrinsically limiting. The details of flow patterns and temperature gradients within the ejector cannot be considered within its framework. It would be possible to increase the sophistication of the one-dimensional model by replacing nozzle and diffuser efficiencies with friction factors and shock calculations. However, given the complexity of the phenomena taking place, this is probably not worthwhile.

The assumption of cross-sectional homogeneity and thermodynamic equilibrium is clearly incorrect. Numerous studies have shown that a saturated liquid expanding through a nozzle is in a non-equilibrium inhomogeneous state. It appears that the flow typically consists of a core of metastable liquid surrounded by an annulus of saturated vapor. The non-equilibrium condition of the fluid leaving the nozzle may have important implications for ejector performance.

The assumption of constant pressure mixing is not particularly limiting. Some performance improvements might be obtained by using constant area mixing or a combination of constant pressure and constant area mixing (Keenan et al, 1950), but they would not be dramatic.

This analysis does not address the problem of off-design performance. In general, ejectors perform poorly away from their design points, so this limitation may be important.

RESULTS

Table 1 compares the performance of an ideal ejector expansion refrigeration cycle (all efficiencies 1) with that of an ideal vapor compression cycle for standard evaporating and condensing temperatures. The Table shows significant increases in COP and decreases in compressor displacement for all refrigerants, but the changes are much larger for some refrigerants than for others. It also shows that the ejector provides a small but significant part of the overall compression ratio in the ejector expansion cycle. In evaluating the results shown in the Table, one must keep in mind that performance improvements decrease with decreasing temperature difference and with decreasing ejector component efficiencies. The standard vapor-compression cycle displacements and COP's shown here match those given in ASHRAE (1985), but vary slightly from those given in ASHRAE (1989) due to a different property formulation. The displacements and COP's shown for R-113 and R-114 are for cycles with wet vapor exiting the compressor.

The improvement in COP with the ejector expansion system varies from refrigerant to refrigerant because the sources of loss in the standard vapor-compression cycle vary. For some refrigerants, such as ammonia, a large part of the loss in the ideal standard cycle is due to heat transfer from the superheated vapor. For such refrigerants the potential increase in COP by reducing the loss in the expansion process is limited. For others, such as R-502, there is little discharge superheat and almost all the ideal standard cycle loss is in the expansion process. For these refrigerants the potential increase in COP with the ejector expansion cycle is much greater.

Table 1 - Ejector Cycle Performance vs. Standard Cycle Performance. -15 C (5 F) Evaporator Temperature; 30 C (86 F) Condenser Temperature; Compressor, Nozzle, Diffuser Efficiencies 1.

Refri-gerant	Ejector Expansion			Standard Vapor-Compression		Ratios Ejector/Standard	
	COP	Displacement $m^3/kJ*10^3$ (CFM/Ton)	Fraction of Compression in Ejector	COP	Displacement $m^3/kJ*10^3$ (CFM/Ton)	COP	Displacement
R-11	5.70	4.03 (30.0)	0.0422	5.03	4.91 (36.5)	1.13	0.82
R-12	5.70	0.623 (4.64)	0.0795	4.70	0.783 (5.83)	1.21	0.80
R-22	5.61	0.385 (2.86)	0.0777	4.66	0.477 (3.55)	1.20	0.81
R-113	5.73	10.26 (76.3)	0.0443	4.90	13.51 (100.6)	1.17	0.76
R-114	5.71	1.978 (14.72)	0.0755	4.60	2.69 (20.0)	1.24	0.74
R-500	5.69	0.528 (3.93)	0.0803	4.68	0.666 (4.96)	1.21	0.79
R-502	5.67	0.357 (2.66)	0.1113	4.35	0.480 (3.57)	1.30	0.74
R-717 (NH_3)	5.33	0.400 (2.98)	0.0408	4.76	0.462 (3.44)	1.12	0.87

The difference in performance for various refrigerants, as well as the effect of reduced ejector component efficiencies, is examined in Figure 3. The COP ratio plotted in Figure 3 is for motive nozzle, suction nozzle, and diffuser having equal efficiency, with that efficiency plotted on the x-axis. COP ratio is seen to vary strongly with efficiency at efficiencies near 1.0, varying less strongly as efficiency decreases. Even for conservative ejector component efficiencies of 70-80%, the ejector expansion cycle offers considerable improvement over the standard vapor-compression cycle. At efficiencies of zero, the COP is the same as that for a standard cycle. Under these conditions the ejector expansion cycle is simply a standard vapor-compression cycle with a separator to remove flashed vapor before the refrigerant enters the evaporator.

Figure 4 shows the actual COP, rather than COP ratio, for various refrigerants. It shows an interesting trend in performance vs efficiency with the halocarbon refrigerants. The halocarbons with the largest COP improvement are those with the lowest COP in the standard vapor-compression cycle. While for an ideal standard cycle these refrigerants vary widely in COP, for the ideal ejector expansion cycle they vary little. Evidently the differences in COP among the halocarbons are almost entirely due to different expansion valve losses. Ammonia has higher COP than most of the halocarbons for the ideal standard cycle but lower COP for the ideal ejector cycle. This is because expansion valve losses are lower than for most of the halocarbons but compressor superheat losses are higher.

Figure 5 shows the effect of varying the evaporator and condenser temperatures for an ideal ejector expansion cycle using R-12. The ratio of ejector expansion COP to standard vapor-compression cycle COP is seen to increase with temperature difference.

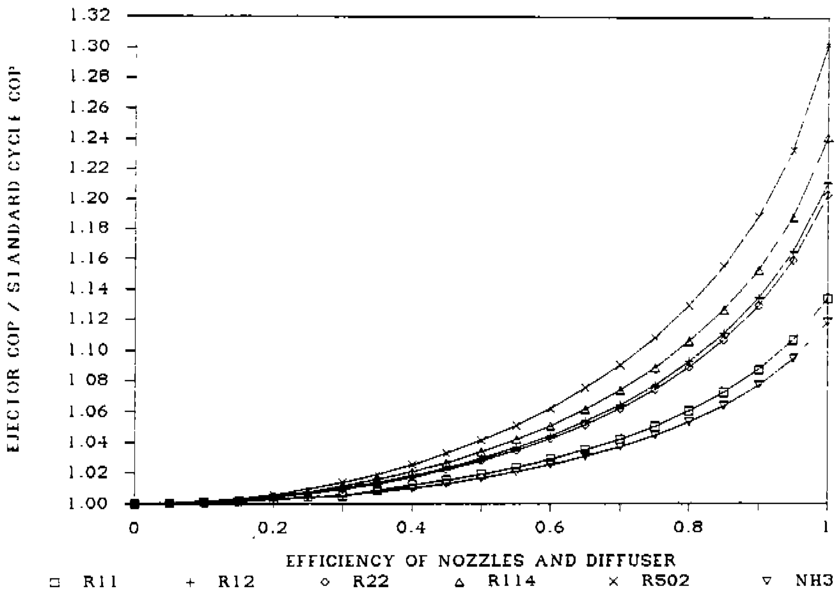

Figure 3. COP Ejector Expansion / Standard Cycle vs Nozzle, Diffuser Efficiency. Various Refrigerants; -15 C (5 F) Evaporator Temperature; 30 C (86 F) Condenser Temperature; Compressor Efficiency 1, other Efficiencies Equal as given.

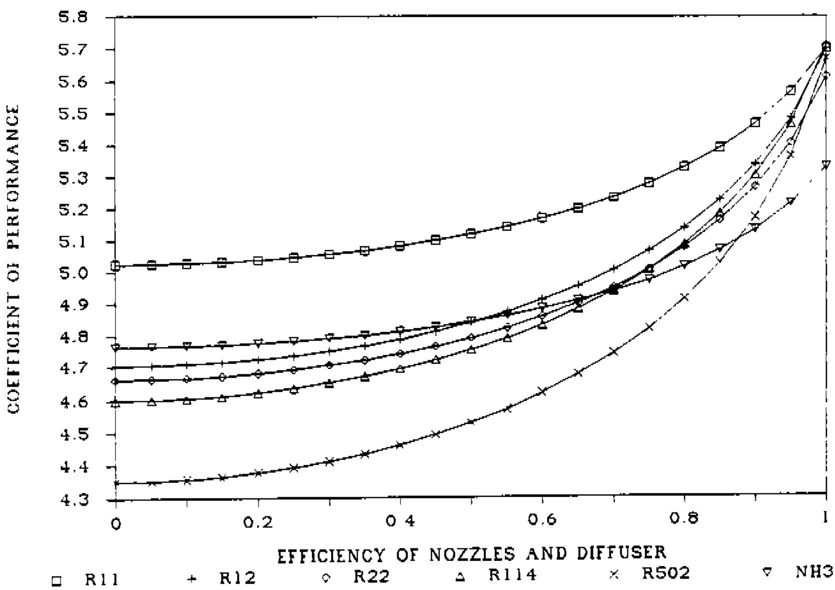

Figure 4. COP of Ejector Expansion Cycle vs Nozzle, Diffuser Efficiency. Various Refrigerants; -15 C (5 F) Evaporator Temperature; 30 C (86 F) Condenser Temperature; Compressor Efficiency 1.0, other Efficiencies Equal as given.

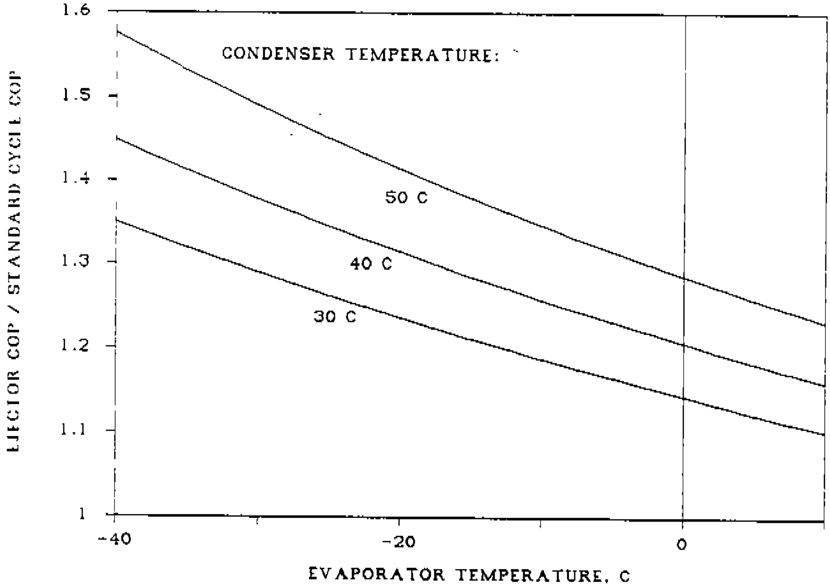

Figure 5. COP Ejector Expansion / Standard Cycle vs Evaporator Temperature. R-12; Various Condenser Temperatures; Nozzle, Diffuser, Compressor Efficiencies 1.

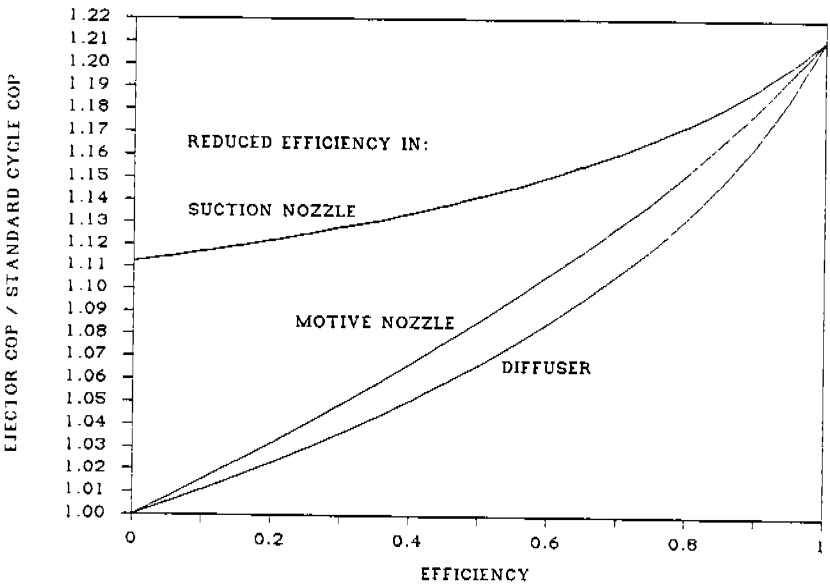

Figure 6. COP Ejector Expansion / Standard Cycle vs Single Component Efficiency. R-12; -15 C (5 F) Evaporator Temperature; 30 C (86 F) Condenser Temperature; Compressor Efficiency 1, other Efficiencies 1 except for Reduced Efficiency given.

Figure 6 shows the effect of inefficiency in one of the ejector components while the other two remain ideal. Lowering the efficiency of the motive nozzle or of the diffuser has approximately the same effect: a steady reduction in COP from that of the ideal ejector expansion cycle to that of the standard vapor compression cycle. Lowering the efficiency of the suction nozzle has a less severe effect, since mixing pressure can be raised to compensate for reduced suction nozzle efficiency. The COP ratio shown for zero suction nozzle efficiency and is the same as the COP ratio shown in Figure 3 for efficiencies of 1.0 and mixing pressure equal to evaporator pressure.

DISCUSSION

Potential of Ejector Expansion Cycle

The ejector expansion cycle shows potential for significant improvement over the standard vapor compression cycle. It offers increased coefficient of performance, decreased compressor displacement, and decreased compression ratio for the same operating conditions. These benefits are obtained by the addition of equipment that is intrinsically durable and low in cost.

There are some important caveats in these apparently rosy predictions. One is that a means must be developed for controlling both cycle capacity and liquid flow within the cycle. The conventional expansion valve control is useless, since it would defeat the performance improvements of the ejector expansion system. Newton (1972a, 1972b) proposed two ways of controlling liquid flow through the ejector. One was to inject small amounts of hot gas into the liquid leaving the condenser, controlling its specific volume and thus the mass flow through the ejector motive nozzle. Another was to build an adjustable ejector, in which motive nozzle efficiency would be maintained while changing motive nozzle area. Newton also proposed using a liquid leg, instead of an expansion valve, to maintain the pressure difference between the ejector discharge and the evaporator. The author does not know to what extent Newton's inventions were developed.

The other major caveat stems from our lack of understanding of rapidly expanding two-phase flows. As stated earlier, the homogenous equilibrium model used in these calculations is known to be inaccurate. Unfortunately, there is no agreement on a really suitable model. The problem has been extensively studied, largely in order to predict the effects of water line breaks in nuclear power plants, but no model effective over a wide range of conditions has emerged. The homogenous equilibrium model is particularly inadequate for the flashing flow in the ejector motive nozzle. Engel (1963) gave efficiencies of 85-98% for motive nozzles in single-phase ejectors. By contrast, Leigh (1970) found that a motive nozzle efficiency of 0.75 gave predictions that best fitted his test data for a two-phase ejector.

Impact on New Refrigerant Research

Should the ejector expansion refrigeration cycle become a practical alternative to the standard vapor-compression cycle, it will change the criteria for optimizing new refrigerants. With the standard cycle, both expansion valve losses and compressor superheat losses have important effects on cycle COP. With the ejector expansion cycle, expansion valve losses are greatly reduced. Potential refrigerants which are unacceptable due to large expansion valve losses in a standard vapor-compression cycle may be much more attractive when used in an ejector expansion cycle.

Directions for Further Work

Further work must be done both experimentally and analytically:

1. Within the limitations of the homogenous equilibrium model, the ejector expansion cycle must be more thoroughly analyzed. Constant area mixing in the ejector, as well as off-design ejector operation, must be studied. Control techniques must be proposed and examined. Noting the difference in COP predictions between ASHRAE (1985) and ASHRAE (1989),

it is important that calculations be done with the most accurate equations of state available. Performance of the cycle with new alternative refrigerants must be calculated.

2. Ejector expansion refrigeration systems must be built and tested, both in order to demonstrate the concept's practically and to uncover deficiencies in analyses.

3. Details of the two-phase flow phenomena within the ejector must be studied experimentally, analytically, and numerically. Such studies are vital to optimizing the ejector expansion system and may also provide a basis for modeling other flashing flows.

Items 1 and 2 are now being pursued at Virginia Polytechnic Institute and State University.

CONCLUSION

The ejector expansion refrigeration cycle offers increased coefficient of performance, decreased compressor displacement, and decreased compression ratio as compared with a standard vapor-compression cycle operating under the same conditions. It may provide significant decrease in operating cost for modest increase in first cost.

The relative COP of refrigerants when used in the ejector expansion cycle is different from the relative COP of the same refrigerants used in the standard vapor-compression cycle. This may impact the search for new, non-CFC, refrigerants.

The processes within the two-phase ejector are poorly understood, and many details of the implementation of the ejector expansion cycle have not yet been worked out. Further research is needed.

REFERENCES

ASHRAE, 1985, *Handbook of Fundamentals*, American Society of Heating, Refrigeration, and Air Conditioning Engineers, pp 16.8-16.9.

ASHRAE, 1989, *Handbook of Fundamentals*, American Society of Heating, Refrigeration, and Air Conditioning Engineers, p 16.7.

Bonnington, S.T. and King, A.L., 1972, *Jet Pumps and Ejectors - A State of the Art Review and Bibliography*, British Hydromechanics Research Association.

Downing, R.C., 1974, "Refrigerant Equations," *ASHRAE Transactions*, Vol 80, Part 2, pp 158-174.

Engel, M.O., 1963, "Some Problems in the Design and Operation of Jet Ejectors," *Proceedings of the Institution of Mechanical Engineers*, Vol 177, pp 347-357.

Flügel, G., 1941, "The Design of Jet Pumps," NACA-TM-982, National Advisory Committee for Aeronautics.

Haar, L. and Gallagher, J.S., 1978, "Thermodynamic Properties of Ammonia," *Journal of Physical Chemistry Reference Data*, Vol 7, pp 635-788.

Keenan, J.H., Neumann, E.P., and Lustwerk, F., 1950, "An Investigation of Ejector Design by Analysis and Experiment," *Journal of Applied Mechanics*, Vol 17, pp 299-309.

Kemper, G.A., Harper, G.F., and Brown, G.A., 1966, "Multiple Phase Ejector Refrigeration System," US Patent Number 3,277,660, assigned to Joseph Kaye & Co., Cambridge, MA.

Leigh, J.H., 1970, "Multiple Phase Ejector Pilot Plant," Research and Development Progress Report No. 578, US Department of the Interior.

Newton, A.B., 1972a, "Capacity Control for Multiple-Phase Ejector Refrigeration Systems," US Patent Number 3,670,519, assigned to Borg-Warner Corporation, Chicago, IL.

Newton, A.B., 1972b, "Controls for Multiple-Phase Ejector Refrigeration Systems," US Patent Number 3,701,264, assigned to Borg-Warner Corporation, Chicago, IL.

Stoeker, W.F., 1958, *Refrigeration and Air Conditioning*, 1st Ed, McGraw-Hill, pp 194-204.